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Potential Application of NASA Aerospace Technology to Ground-Based Power Systems

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POTENTIAL APPLICATION OF NASA AEROSPACE TECHNOLOGY TO GROUND- BASED POWER SYSTEMS

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ABSTRACT

A review of some of the basic gas turbine technology being developed at the NASA John H. Glenn Research Center at Lewis Field, which may have the potential to be applied to ground-based systems, is presented in this paper. Only a sampling of the large number of research activities underway at the Glenn Research Center can be represented here.

The items selected for presentation are those that may lead to increased power and efficiency, reduced cycle design time and cost, improved thermal design, reduced fatigue and fracture, reduced mechanical friction and increased operating margin. The topic of improved material will be presented by Misra et al. in this conference and shall not be discussed here.

The topics selected for presentation are key research activities at the Glenn Center of Excellence on Turbomachinery. These activities should be of interest and utility to this ISABE Special Forum on Aero-Derivative Land-Based Gas Turbines and to the power industry.

INTRODUCTION

A variety of research activities are carried out in a number of disciplines at the NASA Glenn Research Center. These discipline activities include fluid mechanics, heat transfer, combustion, materials, structures, instrumentation, controls and system analysis. In all of these disciplines, a combination of experiments, analyses and computations have been used in a synergistic fashion. The resulting technology has provided benefits to the aircraft engine industry as well as to a large number of other industries; ranging from automobile to vacuum cleaner manufacturers. The applicability of the NASA research to ground and marine power applications is, quite naturally, very direct and has been utilized to a large extent.

This paper will describe a few selected research activities underway at the Center that may have potential application to the ground and marine based power industry. The material is not totally inclusive due to space and time limitations, but is representative of the research with potential applicability. The topics covered are related to increased power and efficiency (optical tip clearance measurement technique), reduced cycle design time and cost (conservation/solution element algorithm), improved thermal design (turbine tip heat transfer), reduced fatigue failures (turbomachinery aeroelasticity analysis), improved power and efficiency (wave rotors) and reduced friction and stall control (magnetic bearings).

MEASUREMENT PROBES

Optical Tip Clearance Probe

One way to increase the power and efficiency of gas turbines is to minimize the loss due to the clearance between the rotor blade tips and the turbine casing. The clearance varies with temperature within the turbine. During startup, the rotor expands faster than the casing and the clearance is small. As the casing temperature rises and it expands, the clearance becomes larger. Higher efficiencies are attained with small clearances. However, tight clearances lead to case rubbing, tip erosion and lower efficiencies. New measurement technology that provides accurate measurement of blade tip clearances during startup, steady operation and

shut-down of the turbine could lead to turbine designs that optimize tip clearances. Therefore, the judicious use of cooling air to control mechanical growth of the casing and rotor could reduce turbine losses.

A promising technique is being investigated at the Glenn Research Center [1,2] using a dual laser beam integrated fiber optic scanning system for direct measurements and monitoring of rotating blade tip clearance. The operating principle, shown in Fig. 1, utilizes two optical probes in a radially traversing unit.

The two beams are focused on the blade tip. As the probe is advanced radially inward, the separation between the two reflected light pulses gradually decreases when a blade is in region I, as shown in the lower portion of Fig. 1. The two pulses gradually merge in the focal region, where pulse 1 and pulse 2 are overlaid. In region II , the pulses again move apart. The probe actuator is positioned to yield a minimum width of the merged signal; i.e. pulse 1 (P1) and pulse 2 (P2) are coincident as shown.

The measurement accuracy is related to the extent of the focal region. For a typical value of 0.005 inch and assuming that the probe can be positioned within 20% of the focal region , the tip clearance resolution can be 0.001 in. The tip clearance at a given operating condition can be deduced from the amount of probe travel from the established position of the probe face relative to the inner edge of the rub strip, and the statically determined focal length.

The advantage of the optical tip clearance probe over non-optical types such as capacitance or eddy current sensors is that it has much higher frequency response time, the blade tip does not have to be metallic, and the clearance can be measured directly as opposed to using calibration. The disadvantage of the optical probe is that it requires actuation.

The optical probe can potentially be used to calibrate other types of sensors, such as eddy current devices. In addition, the optical probe can be used to calibrate the time-of-flight tip clearance probe [1], if high accuracy is desired and the installation of the traversing probe is not feasible.

The width of the focal region is much smaller than the blade width and hence, as the beam coincidence is approached, the receiving fibers from one probe will receive some light transmitted by the other probe in addition to the light transmitted by the particular's probe single-mode fiber. Therefore, some uncertainty will exist regarding the source of the signals. However, it can be seen that in region I, the upper (pressure) surface of the blade will generate the rising slope of the signal from the light emanating strictly from probe P1. Likewise, the suction side of the blade will generate the falling slope of the signal in region I as the blade leaves the region. These pulse edges are therefore defined all the way to the minimum pulse width. Since the cross section of the blade tip as traced by a stationary beam changes as the blade deforms, the minimum pulse width will change with operating conditions; however, it will always be associated with the focal plane.

Research activities underway are initially aimed at integrated design of the actuation mechanism and the probe seal, to be followed by the fabrication of a remotely controlled tip clearance probe system incorporating two laser diodes and the photo detection circuitry. Testing will be performed in a spin facility equipped with a prototype fan blisk.

Subsequent work will involve the development of a feedback control mechanism that will position the probe so that the beam intersection resides on a passing blade tip. Successful achievement of this device will provide a direct measurement of the tip clearance (radial deflection) for every blade. It is envisioned that a chordwise distribution will also be obtainable.

COMPUTATIONAL FLUID DYNAMICS

Space-Time Conservation Element And Solution Element Method (CE/SE)

Higher fidelity computer simulations are being developed in order to reduce design cycle time and cost. As the accuracy of computer calculations are increased, fewer experimental prototypes are required resulting in lower development cost. In addition, innovative ideas such as the recently adopted three-dimensional blade shape design can be explored on the computer. A significant reduction in computational "turnaround time" will also enable three-dimensional viscous calculations to be used more frequently. At the heart of these codes is the key issue of the algorithm or method of solution.

A new algorithm has been developed at the Glenn Research Center [3,4]. It is different from previous conventional numerical methods and is characterized by a fundamentally simple concept; namely the conservation of flux in both space and time. The CE/SE method solves a set of integral equations derived directly from the physical conservation laws. A detailed description and comparison to conventional finite-volume upwind methods is given in Ref. 4. The development of the method was guided by the requirements that (1) both local and global flux conservation in space and time is enforced (space and time are unified and treated on the same basis),

(2) fluxes are truly multi-dimensional and directly evaluated at the interfaces of immediate neighboring cells with no interpolation or dimensional-splitting, and (3) both dependent flow variables and their derivatives are solved simultaneously with no need for extending the computations to a stencil of cells to achieve higher accuracy. It has been demonstrated that the method is sufficiently robust to cover a spectrum of inviscid problems from weak linear acoustic waves to high speed flow with shocks. The CE/SE method is a uniquely new algorithm for solving problems where discontinuities occur. The current state-of-the-art algorithms either are inherently dissipative or require excessive numerical dissipation to ensure stable shock capturing. This may be a serious deficiency when attempting to solve flows with both shocks and acoustic waves present. Numerical dissipation can obscure the acoustic waves, yielding inaccurate results. The CE/SE method minimizes numerical dissipation and, in addition, allows for simple, effective, non-reflecting boundary conditions. The resulting benefits are: reduced implementation time, reduced computing time, and improved accuracy. All these factors are essential for design purposes.

To date, computations have been carried out for shocks, rarefaction waves, vortices, detonation waves and the interaction of an acoustic wave with a vortex. A Navier-Stokes extension of the analysis has been used for computing shock-boundary layer interactions. Other applications of the code to date have included a blade-vortex interaction, that captured the generation of strong acoustic waves as well as demonstrated the effectiveness of the non-reflecting boundary conditions. Low numerical dissipation has been demonstrated, indicating the usefulness of this method for aero-acoustics. The CE/SE method can be formulated to work on unstructured meshes enabling analysis of flow domains with complex geometries.

Currently, extensions to a three-dimensional inviscid version are underway. The critical issues remaining for verification of this method include the demonstration of low numerical dissipation on meshes with widely varying cell size, the continuing extension to viscous and chemically reacting flows. Research is underway to address all these issues.

Turbine Tip Heat Transfer

Turbine blade tips are subjected to high temperatures associated with the leakage flow through blade tip gaps, which may result in burnout and/or severe oxidation. In order to design efficient cooling schemes, it is necessary to have a detailed knowledge of the gap flow field and the tip-heating pattern. CFD methods offer the possibility of determining the effect of selective parametric variations on the tip flow field and the associated thermal loading. Three-dimensional heat transfer computations offer the potential for improved thermal design.

A multi-block version of the code previously developed by Arnone et al.[5] is being currently assessed against available tip data[6]. This code is a general-purpose, finite-volume, Reynolds-averaged Navier-Stokes code using a multi-stage Runge-Kutta based multigrid method, with central differencing and artificial dissipation[4]. A k-ω turbulence model developed by Wilcox and modified by Menter is used. The model integrates to the walls and no wall functions are used. A constant value of turbulent Prandtl number (0.9) and molecular Prandtl number (0.7) are used and molecular viscosity is assumed to be function of temperature to the 0.7 power.

The most recent code validation has been carried out using cascade data. Gridpro™ was used to generate the grids for both a sharp and a rounded tip edge. The number of grid points was about a million and a half; with the first cell center point adjacent to the wall having a non-dimensional distance (y^+) of unity or less. Thirty three points were used in the boundary layers and 65 points were used in the spanwise direction within the tip region. The grid was generated for the experimental cascade of Ref. 6. Computations for both sharp and rounded edges were performed with the resulting streamlines shown in Fig. 2. The experimental heat transfer coefficients are compared with the computed results in Fig. 3. Similar comparisons were made for the sharp edged blade tip with about the same level of agreement. In order to verify the computational method, a great deal more data needs to be obtained.

Our current ability to calculate heat transfer appears to be at a point where it is leading our experimental data acquisition capability, especially for realistic turbine operating conditions. Additional turbine tip data is needed in order to advance our computational accuracy through improved correlations and/or modeling.

Computational Aeroelasticity

In order to reduce turbomachinery fatigue failures, it is necessary to understand the forces that may lead to blade vibrations as well as the response characteristics of rotating blades. Issues of life and durability are critically dependent on the aeroelastic design of the turbomachinery component.

Aeroelasticity deals with interaction among aerodynamic, elastic, and inertial forces. In gas turbine

engines, the flowing gas interacts with the stationary elements and the rotating blades of the turbomachinery. The unsteady interaction between the aerodynamics and the structural dynamics can lead to two important phenomena that cause failure of the structure, namely, flutter and forced vibrations.

Flutter is the self-excited vibration of blades. The phase relationship between the blade vibration and the unsteady force on the blade is crucial in determining whether the aerodynamic force does work on the blade or vice versa. If work is done on the blades, energy is extracted by the blade from the fluid flow resulting in a growth in the magnitude of blade vibrations, or flutter.

Forced Response results from the excitation of the structure at its modal frequency by aerodynamic disturbances. The sources of such excitations can be wakes, shocks, and vorticity associated with upstream blade rows (or struts) or other flow asymmetry, and potential effects due to the downstream blade row. Forced response vibrations could lead to high-cycle fatigue (HCF) failure. Figure 4 shows a typical resonance diagram with intersecting lines of natural and excitation frequencies. Resonant vibrations occur at intersections of these lines, where the excitation frequency matches the natural frequency.

In new engines, flutter problems are usually encountered during development or testing. When flutter vibrations are encountered, they frequently lead to costly re-design and re-testing of turbomachinery components, resulting in delays in development programs. On the other hand, many forced response vibration problems are not detected in the development and testing stage and show up only during operation. These problems lead to increased maintenance costs, increased inspection requirements, and unexpected downtimes due to the grounding of aircraft fleets. To prevent forced response vibration failures, designers frequently make blades thicker, which results in a performance penalty and possibly increased emission. The prevention of aeroelastic problems requires accurate numerical modeling tools. Such tools include a consistent coupling between the unsteady aerodynamics model and the structural dynamics model.

The potential benefits of improved aeroelastic modeling include flutter-free operation of highly loaded turbomachinery stages, reduction of number of stages, operation closer to stall, and thinner and more efficient blading. Each of these items has a potential to contribute to a 1-2% reduced specific fuel consumption (i.e., reduced fuel burned). These potential benefits are the main reason that improved aeroelastic modeling is worth consideration for all turbomachinery applications (ground-based as well as aircraft engine). In addition, improved aeroelastic modeling can lead to reduced design cycle time and cost and reduced maintenance costs. The importance of accurate aeroelastic modeling derives from the ability of the designer to know that the turbomachinery blade designs will not be susceptible to aeroelastic problems throughout the range of operation. This knowledge will ensure that the designs are optimal in terms of performance, and not over-designed because of aeroelastic considerations.

Flutter and forced response are typically modeled as being uncoupled. For flutter calculations, the numerical model is generally restricted to a single blade row and flow unsteadiness due to upstream and downstream blade rows is neglected. In the last decade, significant advances have been made in aeroelastic modeling. Much of the recent research in aeroelasticity has been focussed on the unsteady aerodynamics models. There are two basic approaches to the unsteady aerodynamic modeling. At the Glenn Research Center, both these approaches are being pursued. The first approach solves the non-linear unsteady aerodynamic equations such as the Reynolds-averaged Navier-Stokes or Euler equations in a time-accurate manner. The second approach[7], solves a linearized form of these equations. The linearized forms of the equations are obtained from the non-linear equations when one assumes that the unsteady disturbance is small compared to the mean flow. Efficient solutions of the linearized equations are possible by assuming that the unsteady disturbance is harmonic, which eliminates the explicit time-dependence. For the flutter problem, only the inception of flutter is typically of interest. That is, it is important to know the operating conditions under which flutter onset occurs. The actual details of the transient response in flutter are less important. For this reason, linearized methods are well suited for flutter calculations. For forced response problems, the magnitude of the unsteady disturbance cannot always be assumed to be small, and the linearized approach may not always be applicable. In either case, the computational efficiency of the linearized method makes it most suitable for repetitive design-oriented calculations.

Aeroelastic analysis can be performed using either a frequency domain or a time domain approach. Historically, the unsteady aerodynamics models were based in the frequency domain and thus, it was natural to use the frequency domain approach in aeroelasticity. With the availability of time-accurate CFD-based unsteady aerodynamic analysis, the time domain aeroelastic analysis[8] has become available as an option. The time domain approach allows a nonlinear aeroelastic calculation that simultaneously includes effects of blade vibration and unsteady aerodynamic excitation.

Recently, at the Glenn Research Center, a new advanced flutter code[9] has been developed and is being validated. In this code, flutter modeling is based on a time-accurate nonlinear Reynolds-averaged Navier-Stokes CFD model for a single blade row. The code incorporates the dynamic deformation of the blades and the blade structural dynamics characteristics (namely, vibration frequency and mode shape). Also, at the Glenn Research Center, a new advanced forced response code is being developed and validated. In this code, forced response vibration modeling is based on an unsteady aerodynamic Reynolds-averaged Navier-Stokes CFD code for two blade rows with the blades assumed to be rigid. As an option, a single blade row can also be modeled with the upstream blade row influence represented by a time-varying disturbance (gust) at the inlet boundary. The unsteady forces on a blade row from such analyses are used in a structural analysis along with blade structural dynamics characteristics and aerodynamic damping associated with blade vibration to calculate the resulting dynamic stresses on the blade.

An important area of ongoing and future research in propulsion aeroelasticity at the Glenn Research Center is an effort to improve unsteady aerodynamics modeling. Improvements will include better viscous modeling for separated flows, better turbulence models, improved transition modeling, improved modeling of tip-gap flow and better boundary conditions for computational inlet/exit boundaries. Another area of research that holds promise is in multi-stage aeroelastic modeling. With the increased complexity of unsteady aerodynamics modeling there will be a need to improve the numerical algorithms so that computations can be completed with reasonable computer resources. This is especially important if these advanced aeroelastic analysis methods are to be used in a design-oriented manner, rather than only an analysis to check the final design.

Other areas of ongoing and future research include mistuning which occurs because of small blade-to-blade differences arising either through random manufacturing differences or, possibly, because they were intentionally introduced to obtain improvement in aeroelastic characteristics. Mistuning may have significant impact on blade durability. Providing new methods of damping is especially important in new rotor designs such as blisks where the hub and blades are machined from a single piece of metal, thus eliminating the possibility of introducing damping through blade attachments. Therefore, the area of damping remains an active area of research related to aeroelasticity.

WAVE ROTOR

Wave rotor technology offers the potential to increase the performance levels of gas turbine engines (GTE) significantly, within the constraints of current material temperature limitations. The wave rotor is self-cooled and is aerodynamically compatible with the low corrected flow rates supplied by the core compressors of modern aeropropulsion engines. It can be embedded concentrically (Fig. 5) to increase overall engine pressure ratio by a factor of three and peak temperature by twenty-five percent, without increasing the temperature of the rotating machinery. The thermodynamic benefits offered by wave rotor topping are substantial throughout a wide range of engine classes.

Machine Description

The wave rotor comprises of a tip-shrouded rotor that is surrounded by a stationary casing as shown in Fig. 6. The casing endwalls are penetrated by inlet and outlet ducts that port gases of different pressure and temperature to and from the rotor flow annuli. The rotor hub, tip-shroud, and blade surfaces define rotor passages. Gasdynamic (shock and expansion) waves are initiated as the rotor passages open and close to the ported flows in a timed sequence set by the rotor speed and azimuthal location and extent of the ports. These waves compress and expand the gas as they propagate through the rotor passages. In the simplest configuration, the rotor blades are straight and aligned with the axis of rotation and the net shaft power of the machine is zero. The rotative speed is set by aerodynamic design trades and the corrected tip-speeds are typically low (*e.g.*, 300 ft/s). Although the rotor flow field is inherently unsteady, the port flows are essentially steady and the wave rotor can be closely integrated within other steady flow turbomachinery components. In the GTE topping application, fresh air from an upstream compressor enters the wave rotor through the low pressure inlet port. This air is compressed by shock waves as it traverses the rotor and cools the passage surfaces. The compressed air is discharged at the opposite end of the rotor to an external burner at a pressure typically three times higher than the compressor discharge pressure. The burner exhaust gas reenters the wave rotor through the high pressure inlet port. As it traverses the rotor, the hot gas is expanded, it heats the passage surfaces, and is discharged to a downstream turbine. A brief history of wave rotor and a detailed description of the four-port wave rotor shown in Fig. 6 are given in Ref. 10.

Predicted Thermodynamic Benefits

Temperature-entropy diagrams for untopped (baseline) and wave-rotor-topped engines are shown in Fig. 7. The compressor pressure ratio, the burner energy addition, and the turbine inlet temperature is the same for both engines; however, because heat addition occurs at higher pressures and temperatures in the topped engine, and because the expansion and compression work in the wave rotor are equal, the total pressure into the turbine of the wave rotor topped engine is 15 and 20% higher than that of the untopped engine. The higher availability at the turbine inlet translates directly into increased power and efficiency. For example, cycle studies indicate that the specific power and specific fuel consumption (SFC) of the 400 to 600 hp-class Allison-250 turboshaft engine are enhanced by approximately +18 to +20% and -15 to -16%, respectively. The output power of auxiliary power units, with low (4:1) compressor pressure ratios, can be enhanced by +200% for the same fuel flow rate [11]. Two approaches were considered in the analysis of a 25 MW ground power unit. In the first, the wave rotor topping cycle increased power plant output by 14%, from 25 MW to 28.5 MW, at the same fuel consumption rate. In the second, the turbine inlet temperature of the wave-rotor-topped plant was reduced 103 R (57 K) below that of the baseline plant; the wave-rotor-topped plant power produced the same 25 MW at the derated turbine temperature, while consuming 9.7% less fuel. The reduction in turbine inlet temperature could significantly extend turbine life. Wave turbines, which are wave rotors designed to produce net shaft power, promise even higher performance enhancement levels [12].

Advantages Offered to Gas Turbine Engines

Self-cooling. The rotor passage surfaces are alternatively washed by the relatively low temperature compressor discharge and high temperature burner discharge at frequencies much higher than the material thermal-response-time. The rotor surface remains substantially (*e.g.*, 25%) cooler than the burner discharge. This self-cooling feature thus enables the wave-rotor-topped GTE to operate with hotter peak engine temperatures without increasing turbine inlet temperatures.

Low Rotative Speed. Typical wave rotor corrected tip-speeds are a factor of five or six lower than those of modern turbomachines and the rotors are shrouded. The lower centrifugal stress levels can accommodate higher rotor surface temperatures than acceptable in unshrouded, high tip-speed machines; indeed, ceramic rotors may be a viable design choice.

Low corrected flow. Tolerance-related losses associated with small blading significantly challenge the performance of core compressor aft stages. The wave rotor is well suited to these low corrected-flows because the discharge from the full annulus of the compressor diffuser is ducted to the partial-annular port of the wave rotor. This flow concentration allows for aerodynamically efficient rotor passage geometries. Furthermore, the rotor itself is shrouded so that tip leakage losses are eliminated.

Rapid Transient Response and Stability. The wave rotor responds (gasdynamically) to transients in adjacent components within a couple of rotor revolutions. The fast response is quite independent of its instantaneous rotative speed, in contrast to turbomachinery components that must spool up or down. The fast response has been demonstrated in Brown-Boveri's diesel engines supercharger named Comprex®; engines fitted with the wave rotor respond faster to power demand than do the same vehicles fitted a conventional mechanical supercharger. Past research has suggested that a wave rotor topping unit can enhance the dynamic stability of the gas turbine engine. Calculations [13] showed the wave rotor to be stable and well behaved during fuel and mass flow transients. More recently, a numerical study [14] showed that the wave-rotor-enhanced engine indeed has less tendency to surge during rapid fuel flow changes than do untopped GTEs. This stabilizing feature may allow wave-rotor-topped engine operation at reduced compressor stall margin.

Compact. The wave rotor is compact in the engine topping application because it is introduced aft of the compressor: the rotor length typically varies from one-half to two times the rotor outer-diameter and the rotor outer-diameter is typically about the same as those of the adjacent compressor and turbine.

Critical Issues and Technical Challenges

The wave rotor component poses certain traditional technical challenges due to its inherent unsteadiness and geometry. Additional issues are posed when integrating the wave rotor into the elegantly simple GTE flow path.

Component Design and Analysis. The wave rotor flow field is inherently unsteady and has been historically difficult to analyze. A suite of aerodynamic design/analysis tools and design methods [15-18] have been developed at NASA Glenn to overcome this difficulty. These computational tools have been successfully used to design wave rotor geometries, to map on and off-design wave rotor performance, to analyze wave rotor transients, and to study inherently multi-dimensional

flow features inside the rotor passages. In particular, the one-dimensional design/analysis code has been extensively validated using performance data from NASA Glenn experiments [19,20].

Leakage. Leakage flows between the rotor passages and the plenum between the rotor and casing have been responsible for poor performance in past wave rotor efforts. The leakage flows can be controlled by minimizing the gap between the rotor and the endwall and, when necessary, brush seals can be employed to mitigate leakage [21].

Noise. The gasdynamic waves within the wave rotor generate significant noise. The wave rotor noise issue was successfully addressed by Brown-Boveri by employing randomly pitched rotor blades; however, the noise generated by a wave rotor embedded deep in a GTE remains a research topic.

High Cycle Fatigue. Unsteady pressure loads are established across the rotor blades as the rotor passages open and close to the ported flows. The instantaneous loading across the rotor can be as high as 3:1 at the leading and trailing ends of the rotor. The sudden loading and unloading is experienced by each rotor blades several times per revolution; this can induce high cycle fatigue. The rotor blade design must account for the unsteady loading. Additionally, the potential interaction between the wave rotor and adjacent turbomachinery components must be considered.

Ducting. The wave rotor introduces a new component to the GTE. Flow must be ported to and from the full annuli of the adjacent conventional turbomachinery components to and from the partial sectors of the wave rotor. This ducting constitutes a major modification to the GTE. Indeed the viability of wave rotor integration in a certain GTE may depend upon the posed ducting problems, including the maintainability and life of the additional ducting. Current research includes collaborative work with Rolls-Royce Allison to design aerodynamically efficient transition ducting from the wave rotor to the downstream turbine [22,23]. A wave rotor concept for on-rotor combustion, attempts to eliminate the external burner and associated ducting altogether [24].

Thermal and Mechanical loads. Although wave rotor topping does not increase the temperatures of the rotating machinery, the engine peak temperatures and pressures are increased considerably. These higher temperatures and pressures impose significantly higher thermal and mechanical loads on the hottest components in the cycle. The associated weight penalty is of particular significance to aeropropulsion engines but may be less important for ground power units.

High-Pressure-Turbine Cooling Air. The wave rotor topping introduces an interesting problem: the compressor discharge has insufficient pressure to provide cooling to the first stage of the turbine; therefore, the first stage cooling air must be extracted from the wave rotor topping unit. This high pressure gas is commensurately hotter and therefore more of it is required for the same cooling requirements.

Turning the Rotor. As mentioned, optimum wave rotor corrected tip-speeds are five to six times lower than those of the adjacent turbomachinery. Optimized wave rotors typically are of approximately the same diameter of the upstream and downstream turbomachinery; therefore, the wave rotor must spin at a much lower rotative speed than that of the conventional gas generator. In a power plant application, the rotor would more likely be self-driven. In this approach, the inlet ports and guide vanes would provide the preswirl needed to overcome windage and bearing losses. Numerical simulations suggest this is a practicable approach in terms of resulting wave rotor performance. Alternatively, some type of mechanical lashing to the high spool shaft can be envisaged because the wave rotor spins slower than the adjacent turbomachinery by a constant fraction (1/3) over the full topped-engine operating range [25].

MAGNETIC BEARINGS

Magnetic bearings have the potential to eliminate friction, reduce vibration of the engine shafts and possibly improve the compressor stall margin.

This type of bearing can support a turbomachinery shaft without any physical contact, eliminating mechanical friction and wear. Consequently no lubrication system is required for the machine, eliminating a major area of maintenance and failure. For this reason one of the first commercial application areas was in compressors for natural gas pipelines in remote areas of Canada, where low maintenance is important.

The basic operating principle is rather simple. Electromagnets are arranged around a magnetic sleeve on the shaft. Position sensors signal any lateral motion of the shaft off the intended center line. A feedback control system then sends more electric current through the windings of the appropriate electromagnet to pull the shaft back to center. Since this feedback system has very fast response (less than milliseconds), the forces exerted by the bearing can not only center the shaft but can also counteract shaft vibrations, both rigid-body modes (bouncing and tilting) and shaft flexible modes.

The bearing just described is a radial bearing, but an axial, or thrust, bearing is also used. It works on a similar principle but has a different and simpler shape. The electric power required by a magnetic bearing sized for a turbine engine is only a few hundred watts, and the total losses of the magnetic bearing system are far less than the power expended in conventional bearings due to viscous drag in the bearing and in the oil pumping and scavenging systems.

A magnetic bearing is considerably larger than a rolling element bearing. However, if we consider the entire system including all auxiliary components (e.g., controller and power amplifiers for magnetic bearings and pumps, coolers, sumps and piping for rolling element bearings), the magnetic system is expected to be smaller and lighter than the rolling element system in most turbomachines. In a turbine engine there are typically two concentric shafts, and two or more radial bearings and one thrust bearing are used on each shaft. Fig. 8 indicates where two of the radial magnetic bearings would be installed on one shaft in an engine. A back-up or "catcher" bearing with an air gap is also used next to each magnetic bearing to catch the shaft in case of power loss or other magnetic bearing failure.

The major advantages of magnetic bearings in turbine engines include the elimination of the bulky and failure-prone lubrication system, reduction of shaft vibration, better shaft centering, higher rotational speeds, reduced blade-tip and gas-seal wear, inherent rotor health monitoring from the bearing feedback system and in-situ "balancing" of the rotor. Higher performance can be obtained because of the reduced torque drag on the rotor, maintenance of better blade-tip and seal clearance, lower compressor bleed for bearing cooling and no bleed for oil seals. Greater safety results from the absence of the oil system and its many possible failure points.

There is a possibility that the magnetic bearings can further increase performance by contributing to compressor stall avoidance. This would be done by very small amplitude whirling of the shaft in response to incipient stall gas-pressure fluctuations, thereby changing blade-tip leakage patterns to counteract those pressure fluctuations and avoid stall.

Magnetic bearings are gaining acceptance in more and more rotating machinery applications. Besides the turbocompressor application already mentioned, one of the earliest applications was in turbomolecular vacuum pumps. The absence of oil contamination of the vacuum system is important there, as well as long life at high shaft speeds. The majority of turbomolecular pumps are now sold with magnetic rather than conventional bearings. In the last several years many other commercial applications have occurred, usually in one-of-a-kind installations meant for trial evaluation. These include bearings for high-speed machining spindles and for turbocompressors or turboexpanders in petrochemical processing plants.

Glenn Research Center and others are working on magnetic bearings for aircraft turbine engines and for high-speed energy storage flywheels for space power systems. In the latter application, magnetic bearings are absolutely essential because the flywheels must operate in a vacuum (making conventional high-speed bearings infeasible), and flywheels cannot tolerate the higher level of continuous drain of stored energy that would come from conventional bearing friction.

Magnetic bearings with their associated controllers are no more complex than modern speed- or torque-controlled motors, which are very reliable and long-lived. A large turbomachine with magnetic bearings will be less complex than one with oil-lubricated bearings, with far fewer external components, enabling easier and cheaper maintenance. For advanced aircraft turbine engines in the 2010 timeframe, there are a number of development areas required to bring magnetic bearings to an acceptable technology readiness level. The expected high bearing-compartment temperatures on the order of 1000°F, which virtually rule out conventional bearings, can be tolerated by magnetic bearings, but require the use of oxidation-resistant wire coatings and ceramic insulation. The insulation, in particular, requires more development.

For ground power, the push to ever-higher temperatures may not be so fast as for flight applications, but the goal of higher cycle efficiency will always push in that direction. Glenn Research Center has a new High Temperature Magnetic Bearing Rig for research in this area. Fault tolerance is another active area. For flight, the major motivator is safety of people and protection of expensive equipment. For ground power, the safety issue is different and perhaps easier to satisfy, but redundancy and other fault-tolerant features are still required. Extra coils and power amplifiers are typically used to increase fault-tolerance and to allow continued operation even with some level of failure. Glenn Research Center has a new Fault Tolerant Magnetic Bearing Rig to explore the best use of these and other approaches. The catcher bearing is also part of the fault-tolerance picture. In a ground system, it would come into play only if the other fault-tolerance features cannot support the rotor (i.e. with multiple failures or general power failure). The type of catcher bearing to be used depends on the application and the likelihood of failure, as well as the consequence or cost of a failure. For ground power applications this requirement is less

severe than for a single-engine fighter aircraft that would need to fly to a base after a failure. The catcher bearing for ground power machines might be as simple as a dry-lubricated sleeve bearing (with a gap).

Some of the difficulties in applying magnetic bearings to turbine engines are easier to solve for a power-generating ground derivative engine than for an aircraft engine. For example, there are no maneuver centrifugal loads or gyroscopic loads in the ground case, leading to smaller bearings. The fault tolerance requirements may be easier to satisfy. Weight is not directly an issue. For these and other reasons it appears that magnetic bearings might be introduced into ground derivative engines before they are used in flight engines.

On the other hand, low capital cost may be more important for ground power systems and longer operating life may be required. Certainly, there are differences in the engine operating conditions. The ground engine operates at higher ambient pressure but without inlet ram air. It is attached to a generator instead of a fan, and it probably has a more constant speed. Other differences may pose unique problems that will not be solved by aircraft-focused research and development. In any case, the two application areas should complement each other, with ground systems benefiting, as in the past, from aircraft engine development, and aircraft systems probably benefiting from the introduction of magnetic bearings into ground-based systems first.

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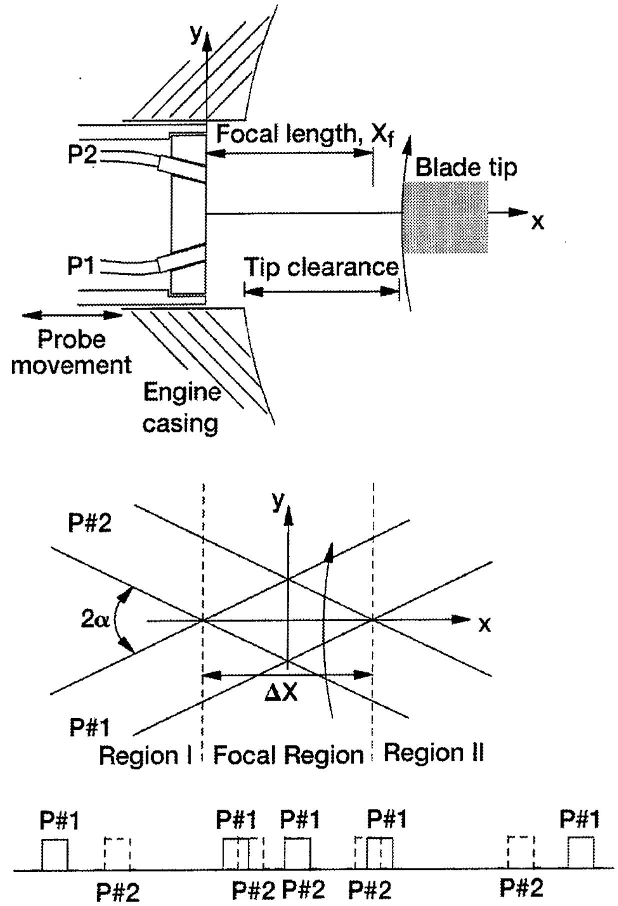
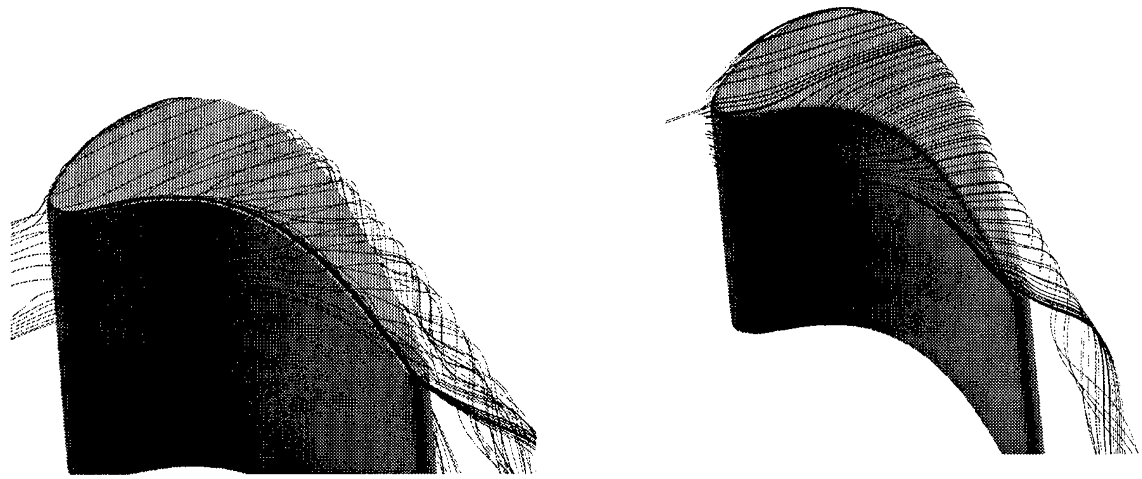
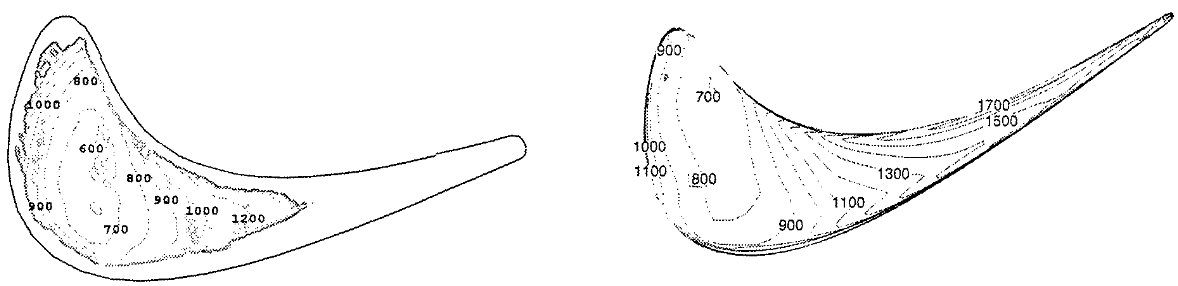


Figure 1: Schematic of a tip clearance probe and voltage pulses at various radial distances.



(a) (b)

Figure 2: Flow streamlines over the blade tip for (a) sharp edge blade and (b) rounded edge blade, (Ref. 6).



(a) (b)

Figure 3: Rounded edge blade tip heat transfer coefficient (W/m^2/K) for 2.03 mm clearance and Tu=5%, (a) measure and (b) calculated, (Ref. 6).

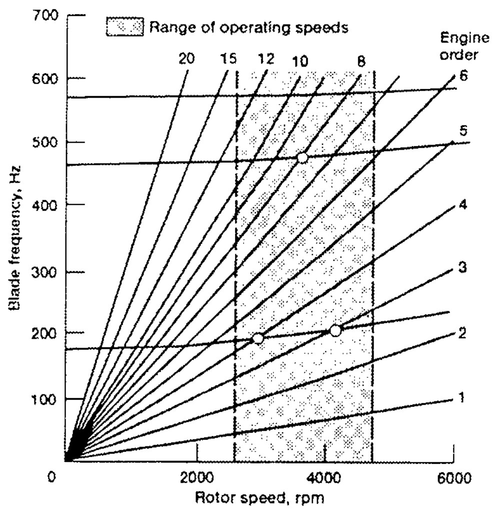


Figure 4: Typical resonance diagram for rotor blade showing possible forced response conditions denoted by circles.

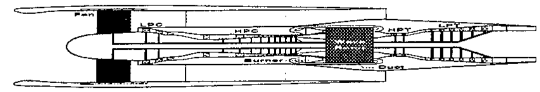


Figure 5: Wave-rotor-topped turbofan engine.

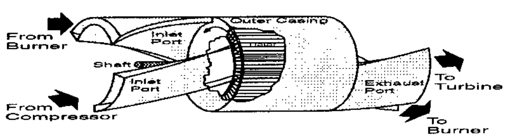


Figure 6: Four-port wave rotor schematic diagram.

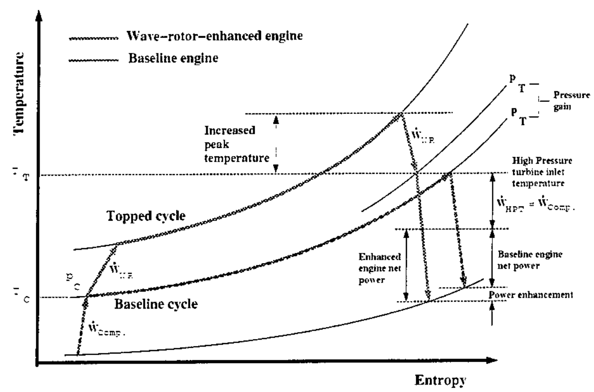


Figure 7: Temperature-entropy diagram showing thermodynamic benefit of wave rotor topping.

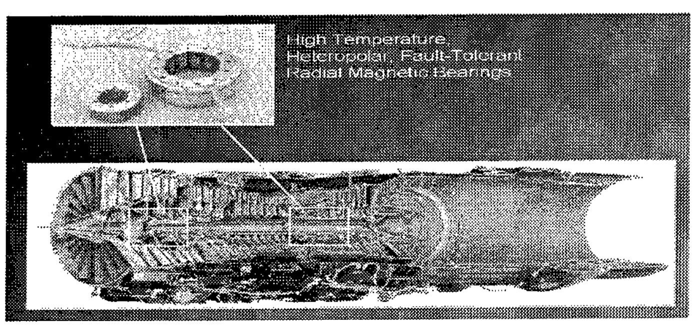


Figure 8: Two potential magnetic bearing locations in a military engine.

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13. ABSTRACT ***(Maximum 200 words)***

A review of some of the basic gas turbine technology being developed at the NASA John H. Glenn Research Center at Lewis Field, which may have the potential to be applied to ground-based systems, is presented in this paper. Only a sampling of the large number of research activities underway at the Glenn Research Center can be represented here. The items selected for presentation are those that may lead to increased power and efficiency, reduced cycle design time and cost, improved thermal design, reduced fatigue and fracture, reduced mechanical friction and increased operating margin. The topic of improved material will be presented by Misra et al. in this conference and shall not be discussed here. The topics selected for presentation are key research activities at the Glenn Center of Excellence on Turbomachinery. These activities should be of interest and utility to this ISABE Special Forum on Aero-Derivative Land-Based Gas Turbines and to the power industry.

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